DAMPING OF COMPOSITE MATERIAL STRUCTURES WITH RIVETED JOINTS

A thesis submitted in partial fulfillment of the requirements for a degree of

Bachelor of Technology

in

Mechanical Engineering

by

NIKHIL CHINTHAPATLA

Roll No. : 108ME053

Department of Mechanical Engineering

National Institute of Technology, Rourkela

2012

DAMPING OF COMPOSITE MATERIAL STRUCTURES WITH RIVETED JOINTS

A thesis submitted in partial fulfillment of the requirements for a degree of

Bachelor of Technology

in

Mechanical Engineering

by

NIKHIL CHINTHAPATLA

Roll No. : 108ME053

Under the guidance of

Prof. B. K. NANDA

Department of Mechanical Engineering

National Institute of Technology, Rourkela

2012

National Institute of Technology Rourkela

CERTIFICATE

This is to certify that the thesis entitled ***"DAMPING OF COMPOSITE MATERIAL STRUCTURES WITH RIVETED JOINTS"*** submitted by Mr. **NIKHIL CHINTHAPATLA** in partial fulfillment of the requirements for the award of Bachelor of Technology Degree in Mechanical Engineering at National Institute of Technology, Rourkela (Deemed University) is an authentic work carried out by him under my guidance.

To the best of my knowledge the matter embodied in the thesis has not been submitted to any University/Institute for the award of any Degree or Diploma

Prof. B.K. NANDA

Department of Mechanical Engineering

National Institute of Technology

Rourkela – 769008

DATE:

ACKNOWLEDGEMENT

I express my deep sense of gratitude and reverence to my thesis supervisor **Prof. B. K. Nanda**, Professor, Mechanical Engineering Department, National Institute of Technology, Rourkela, for his invaluable encouragement, helpful suggestions and supervision throughout the course of this work and providing valuable department facilities.

I would like to thank **Prof. K. P. Maity**, Head of the Department, **Prof. C. K. Biswas,** Chairman of Production Engineering, **Prof. S. K. Sahoo** and **Prof. R. K. Behera**, faculty Adviser.

DATE: **NIKHIL CHINTHAPATLA**

108ME053

ABSTRACT

Vibration and noise reduction are crucial in maintaining high performance level and prolonging the useful life of machinery, automobiles, aerodynamic and spacecraft structures. It is observed that damping in materials occur due to energy release due to micro-slips along frictional interfaces and due to varying strain regions and interaction between the metals. But it was found that the damping effect in metals is quite small that it can be neglected. Damping in metals is due to the micro-slips along frictional interfaces. Composites, however, have better damping properties than structural metals and cannot be neglected. Typically, the range of composite damping begins where the best damped metal stops.

In the present work, theoretical analysis was done on various polymer matrix composite (glass fibre polyesters) with riveted joints by varying initial conditions. Strain energy loss was calculated to calculate the damping in composites. Using FEA model, load variation w.r.t time was observed and the strain energy loss calculated was utilised in finding the material damping for Carbon fibre epoxy with riveted joints. Various simulations were performed in ANSYS and these results were utilised to calculate the loss factor, Rayleigh's damping constants and logarithmic decrement.

These results can be used in designing machine tools, aircrafts, spacecraft's, satellites, missile systems and automobiles effectively to maximise the damping capacity and to improve their performances and the product life.

CONTENTS

01. COVER PAGES i - iii

02. ACKNOWLEDGEMENT iv

03. ABSTRACT v

04. LIST OF TABLES AND FIGURES vii

05. INTRODUCTION 01

1.1 BACKGROUND
1.2 OBJECTIVE OF THE WORK

06. LITERATURE REVIEW 03

2.1 INTRODUCTION
2.2 OVERVIEW ON DAMPING
2.3 PAST RESEARCH ON DAMPING

07. COMPOSITES 07

08. DAMPING 09

4.1 DEFINITON OF DAMPING
4.2 TYPES OF DAMPING
4.2.1 MATERIAL DAMPING
4.2.1.1 ENERGY BALANCE APPROACH
4.2.2 FLUID DAMPING
4.2.3 STRUCTURAL DAMPING
4.3 DAMPING MECHANISM IN COMPOSITE MATERIALS

09. VISCO ELASTIC DAMPING 15

5.1 FACTORS AFFECTING VISCO ELASTIC DAMPING
5.2 MATERIALS UTILISED

10. MATHEMATICAL MODELLING 17

6.1 STRUCTURAL DAMPING FACTOR
6.2 COMPLEX STIFFNESS

11. MODELLING AND ANALYSIS OF COMPOSITES WITH RIVETS 19

7.1 MODELLING
7.2 ANALYSIS OF MODEL IN ANSYS
7.2.1 GENERAL OVERVIEW OF DAMPING IN ANSYS
7.2.2 RAYLEIGH DAMPING

12. FINITE ELEMENTAL ANALYSIS 23

8.1 MODAL ANALYSIS
8.2 HARMONIC RESPONSE ANALYSIS
8.3 TRANSIENT DYNAMIC ANALYSIS

13. RESULTS AND CONCLUSION 29

14. REFERENCES 30

LIST OF TABLES AND FIGURES

TABLES:

01. PROPERTIES OF CARBON-FIBRE EPOXY COMPOSITE 16
02. FREQUENCY OUTPUT OF MODAL ANALYSIS 24
03. MAXIMUM AND MINIMUM OF DEFORMATION WITH TIME 27

FIGURES:

01. (Fig 3.1) TYPES OF FIBRE REINFORCED MATERIALS 07
02. (Fig 4.1) MASS-SPRING DAMPER SYSTEM 09
03. (Fig 4.2) RVE MODEL 11
04. (Fig 7.1) MODEL DESIGNED ON CATIA V5R17 19
05. (Fig 7.2) VARIOUS VIEWS OF DRAFTING 19
06. (Fig 7.3) RAYLEIGH DAMPING 22
07. (Fig 8.1) ANSYS MODEL 23
08. (Fig 8.2) MODE Vs. FREQUENCY GRAPH 24
09. (Fig 8.3) MODAL ANALYSIS 25
10. (Fig 8.4) HARMONIC RESPONSE ANALYSIS 25
11. (Fig 8.5) AMPLITUDE Vs. FREQUENCY GRAPH 26
12. (Fig 8.6) PHASE ANGLE Vs. FREQUENCY GRAPH 26
13. (Fig 8.7) MAXIMUM DEFORMATION Vs. TIME GRAPH 28
14. (Fig 8.8) MINIMUM DEFORMATION Vs. TIME GRAPH 28

CHAPTER 1
INTRODUCTION

1.1 BACKGROUND

Damping capacity is an extent of a material's ability to dissipate elastic-strain energy during mechanical vibration or wave propagation. Complications involving vibration arise in many regions of mechanical, civil and aerospace engineering. The damping of a structural component or element is often a significantly overlooked criterion for good mechanical design. Numerous mechanical failures over a seemingly infinite multitude of structures occurred due to lack of damping in structural elements. For accounting the damping effects in a structural material, lots of researches and studies have been done in the field to suppress the vibration and to minimize the mechanical failures.

Since it was found that damping materials can be utilised in treatment in passive damping technology to mechanical components and structures to increase the damping performance, there had been a commotion on the on-going research and studies over the last few periods to either alter the existing materials and components, or to develop an entirely new type of material to improve the structural dynamics of components for which damping concept could be applied. Composite structures are generally polymers, which give various ranges of different compositions which result in different material properties as well as behaviour. Hence, composite damping structures and materials can be developed and tailored quite efficiently for a specific purpose and application.

Problems involving vibration and damping occur in many regions of mechanical, civil and aerospace engineering. Engineering composite structures and materials are generally fabricated using a variety of connections which include bolted, riveted, welded and bonded joints etc. The dynamics of mechanical joints is a topic of special importance due to their strong effect on the performance of the structural material. Moreover, the inclusion of the above mentioned joints play a significant role in the overall system behaviour, particularly the damping level of the components and structures. However, determining damping either by analysis or by experiment is never easy and straightforward keeping in view of the complexity of the dynamic interaction of components. The estimation of damping in beam-like structures using passive damping approach is very essential in problem solving addressed by the present research

1.2 OBJECTIVE OF THE WORK

This thesis provides a final summary of the progress made over the past year on the study of damping of composites with riveted joints, specifically applied to high stiffness and damping structural members. Composite materials are materials which dissipate strain energy when deformed in shear. This technology has a wide variety of engineering applications, including bridges, engine mounts, and machine components such as rotating shafts, component vibration isolation, novel spring designs which incorporate damping without the use of traditional dashpots or shock absorbers, and structural supports.

The main focus of this dissertation is to study the complex behaviour of the composite (viscoelastic) materials, to predict damping effects using method of passive viscoelastic constrained layer damping technology and to show the nature of response of structures using finite element method.

CHAPTER 2
LITERATURE REVIEW

2.1 INTRODUCTION

The widespread application of composite materials in the application of aerospace and sciences have inspired many scientists to study numerous aspects of their structural behaviour. These materials are chiefly utilised in circumstances where a huge strength-to-load ratio is necessary. Likewise to isotropic materials, composite structures and materials are exposed to various types of damage, mostly cracking and de-lamination. The result in alteration of dynamic characteristics and consequently vary the toughness of elements.

Many engineering assemblies are constructed by joining structural constituents through mechanical links. Such assembled structures need sufficient damping to limit excessive vibrations under dynamic loads. Damping in such structures mainly originates from two sources. One is the internal or material damping which is inherently low [1] and the other one is the structural damping due to joints [2].

The latter one offers an excellent source of energy release, thereby adequately compensating the low material damping of structures. But, this is only in case of metallic structures and not in composites. It is estimated that metallic structures consisting of bolted or riveted members contribute about 90% of the damping through the joints [3]. The internal damping or material damping in case of composites is generally more, when compared to material damping in metallic structures. Often, damping in composites starts when the best damped metal stops. For this very reason, damping in composites is of recent interest and many researches are being done.

2.2 OVERVIEW ON DAMPING

The 3 crucial factors that determine the dynamic responses of a structural material and its noise propagation features are mass, rigidity and damping. Mass and rigidity are associated with storage of energy. Damping results in the release of energy by a vibration system. For a linear system, when the forcing frequency is the equal to the natural frequency of the system, the response is very huge and can easily cause hazardous consequences. In the frequency domain, the response near the natural frequency is "controlled damping". Larger damping can help to decrease the amplitude of resonating structures. Increased damping also results in faster deterioration of free vibration, reduced dynamic stresses, smaller structural response to noise and sound, and increased noise propagation loss above the threshold frequency. A lot of literatures have been published on damping due to vibration. ASME published a collection of papers on structural damping in 1959 [6]. Lazan's book published in 1968 gave a very good idea on damping research work, discussed different mechanisms and forms of damping, and studied damping at both the microscopic and macroscopic levels [7]. Lazan conducted comprehensive studies into the general nature of material damping and presented damping results data for almost 2000 materials and test conditions. Lazan's results show that the logarithmic decrement values increase with dynamic stress, i.e., with vibration amplitude, where material damping is the dominant mechanism. This book is also valuable as a handbook because it contains more than 50 pages of data on damping properties of various materials, including metals, alloys polymers, composites, glass, stone, natural crystals, particle-type materials, and fluids. About 20 years later, Nashif, Jones and Henderson published another comprehensive book on vibration damping [8]. Jones himself wrote a handbook especially on viscoelastic damping 15 years later [9]. Sun and Lu's book published in 1995 presents recent research accomplishments on vibration damping in beams, plates, rings, and shells [10]. Finite element models on damping treatment are also summarized in this book. There is also other good literature available on vibration damping [11-13].Damping in vibrating mechanical systems has been subdivided into two classes: Material damping and system damping, depending on the main routes of energy release. Coulomb (1784) postulated that material damping arises due to interfacial friction between the grain boundaries of the material under dynamic condition. Further studies on material damping have been made by Robertson and Yorgiadis (1946), Demer (1956), Lazan (1968) and Birchak (1977). System damping arises from slip and other boundary shear effects at mating surfaces, interfaces or joints between distinguishable parts. Murty (1971) established that the energy released at the support is very small compared to material damping.

2.3 REVIEW OF PAST RESEARCH ON DAMPING OF COMPOSITE MATERIALS

Bert [14] and Nashif et al.[15] had done survey on the damping capacity of fibre reinforced composites and found out that composite materials generally exhibit higher damping than structural metallic materials. Chandra et al. [16] has done research on damping in fibre-reinforced composite materials.

Composite damping mechanisms and methodology applicable to damping analysis is described and had presented damping studies involving macro-mechanical, micromechanical and Viscoelastic approaches. Gibson et al.[17] and Sun et al.[18,19] assumed viscoelasticity to describe the behaviour of material damping of composites.

The concept of specific damping capacity (SDC) was adopted in the damped vibration analysis by Adams and his co-workers [20-21], Morison [22] and Kinra et al [23].

The concept of damping in terms of strain energy was apparently first introduced by Ungar et.al [24] and was later applied to finite element analysis by Johnson et.al [25]. Gibson et.al [26] has developed a technique for measuring material damping in specimens under forced flexural vibration. Suarez et al [27] has utilised Random and Impulse Techniques for Measurement of Damping in Composite Materials. The random and impulse techniques utilize the frequency-domain transfer function of a material specimen under random and impulsive excitation. Gibson et al [28] utilised the modal vibration response measurements to characterize, quickly and accurately the mechanical properties of fibre-reinforced composite materials and structures.

Lin et al. [29] predicted SDC in composites under flexural vibration using finite element method based on modal strain energy (MSE) method considering only two inter laminar stresses and neglecting transverse stress.

Koo KN et al. [30] studied the effects of transverse shear deformation on the modal loss factors as well as the natural frequencies of composite laminated plates by using the finite element method based on the shear deformable plate theory.

SINGH S. P et al. [31] analysed damped free vibrations of composite shells using a first order shear deformation theory in which one assumes a uniform distribution of the transverse shear across the thickness, compensated with a correction factor.

Polymeric materials are widely utilised for sound and vibration damping. One of the more notable properties of these materials, besides the high damping ability, is the strong frequency dependence of dynamic properties; both the dynamic modulus of elasticity and the damping characterized by the loss factor [30-35].

Mycklestad [32] was one of the pioneering scientists into the investigation of complex modulus behaviour of viscoelastic materials (Jones, 2001, Sun, 1995). Viscoelastic material properties are generally modelled in the complex domain because of the nature of visco-elasticity. Viscoelastic materials possess both elastic and viscous properties. The typical behaviour is that the dynamic modulus increases monotonically with the increase of frequency and the loss factor exhibits a wide peak [8, 33].

It is rare that the loss factor peak, plotted against logarithmic frequency, is symmetrical with respect to the peak maximum, especially if a wide frequency range is considered. The experiments usually reveal that the peak broadens at high frequencies. In addition to this, the experimental data on some polymeric damping materials at very high frequencies, far from the peak centre, show that the loss factor–frequency curve ‘‘flattens’’ and seems to approach a limit value, while the dynamic modulus exhibits a weak monotonic increase at these frequencies [34-38]. These phenomena can be seen in the experimental data published by Madigosky and Lee [34], Rogers [35] and Capps [36] for polyurethanes, and moreover by Fowler [37], Nashif and Lewis [38] for other polymeric damping materials.

The computerized methods of acoustical and vibration calculus require the mathematical form of frequency dependences of dynamic properties. A reasonable method of describing the frequency dependences is to find a good material model fitting the experimental data.

CHAPTER 3
COMPOSITES

Composite materials are naturally occurring materials or synthetically prepared from 2 or more constituent materials with considerably different physical or chemical properties or both which remain isolated and dissimilar at the macroscopic or microscopic scale within the completed structure. The elements are assorted in such a way so that they can retain their distinctive physical state and which are not solvable with each other nor a new chemical compound is formed. One element is known as reinforcing state which is embedded in another phase called matrix. The most visible applications are pavement in roadways in the form of either steel and aggregate reinforced Portland cement or asphalt concrete.

Most of the fibres are utilised as the reinforcing state and are even tougher than the matrix and this matrix is utilised in holding the fibres intact. Examples: Aluminium's matrix implanted in boron fibres and an epoxy matrix implanted with glass or carbon fibres. These fibres may be long or short, directionally aligned or randomly orientated, or some sort of mixture, depending on the intended use of the material. Commonly utilised materials for the matrix are polymers, metals, ceramics, carbon and fibres are carbon (graphite) fibres, aramid fibres and boron fibres.

Fibre-reinforced composite materials are further classified into the following:

a) Continuous reinforced fibre.

b) Discontinuous reinforced aligned fibre.

c) Discontinuous fibre-reinforced random oriented.

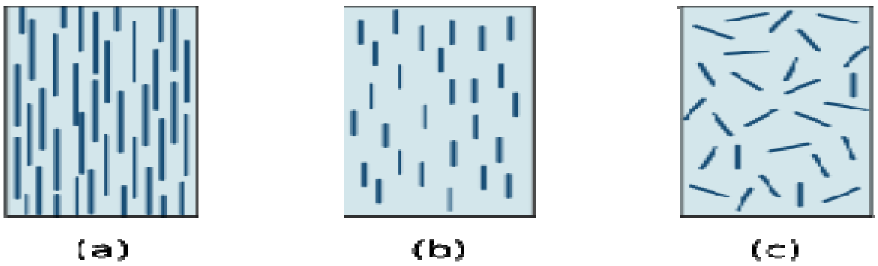


Fig 3.1 Types of Fibre Reinforced Materials

Composites utilised in this work are Carbon epoxy fibre.

Carbon fibre is made up of extremely thin fibres of carbon. It is utilised as an reinforcing agent for many polymer products; the resulting composite material is commonly known as Carbon fibre epoxy. Uses for regular carbon-fibre include applications in the fields of automotive engineering and also aerospace engineering, like Formula One. The toughest and most costly of these essences, carbon nanotubes, are enclosed in some principally polymer baseball bats, car parts and also golf clubs where economically they are available.

Epoxy is a polymer used for thermosetting which is formed by reaction of an epoxide "resin" with polyamine "hardener". Epoxy has a widespread variety of applications, including fibre-reinforced plastic materials and universal purpose adhesives. The uses for epoxy materials are for outer layers which include adhesives, coatings and materials using such composite as those using carbon fibre and fibreglass reinforcements (although polyester, vinyl ester, and other thermosetting resins are generally utilised for glass-reinforced plastic).

The damping rising due to the interactions in-between fibres and matrix can be very huge and are very complex in nature because of many properties of composites which affect the interactions. For example, length, fibre orientation, and interface all affect the damping properties. But the effect of length on damping can be neglected, since it is very small. Damping is generally more when the orientation of fibres is off the axis by 5 to 30 degrees.

CHAPTER 4

DAMPING

4.1 DEFINITION OF DAMPING

In physics, damping is a phenomenon in which the amplitude of an oscillation tends to reduce after every cycle in an oscillatory motion, particularly in case of harmonic oscillator. Friction is generally considered as one such damping effect. In engineering terms, damping can be mathematically modelled as any force which is in sync with the velocity of object and opposite in direction to it. If such a force is proportional to the speed or velocity, as for a simple mechanical gelatinous damper, the force F may be related to the velocity v *given* by F= -cv, where c is the viscous damping coefficient (N-s/m).

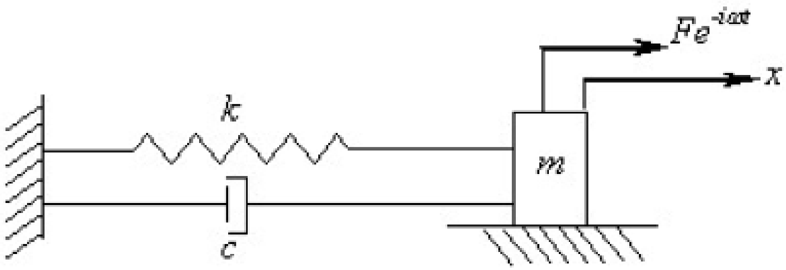


Fig 4.1 Mass and spring damping system

An ideal mass and spring damping system with mass m (kg), viscous damper of damping coefficient c (in N-s/ m or kg/s) and spring constant k (N/m) is subjected to an oscillatory vibration or force then the damping force is given by,

$$F_s = -kx \qquad F_d = -cv = -c\frac{dx}{dt} = -c\dot{x}$$

By applying the Newton's second law, the total force (Ftot) on the body is given by,

$$F_{tot} = ma = m\frac{d^2x}{dt^2}$$

Since $F_{tot} = F_s + F_d$,

then => $m\ddot{x} = -kx - c\dot{x}$

This differential equation can be rearranged as:

$$\ddot{x} + \frac{c}{m}\dot{x} + \frac{k}{m}x = 0,$$

$$\zeta = \frac{c}{2\sqrt{km}}, \quad \omega_0 = \sqrt{\frac{k}{m}}.$$

where ω_0, is the un-damped natural frequency of the vibratory system and ζ, is known as the damping ratio of the spring.

Depending upon the value of ζ, the motion of mass shown in the above Figure can be divided into the following three cases given below:

(1) Oscillatory motion when $0.1<\zeta$;

(2) Non Oscillatory motion when $0.1>\zeta$ and

(3) Critical damped motion when $0.1=\zeta$. In last case, the general solution of the system is $x = (A + Bt)e^{-\omega_0 t}$.

Viscous damping can be utilised whatever may be the form of the excitation. The viscous damping is the Rayleigh-type damping given by

$c = \alpha m + \beta k$

4.2 TYPES OF DAMPING

Three main types of damping are present in any mechanical system:

1) Internal damping (Damping due to material properties)

2) Structural damping (Damping at joints and interfaces)

3) Fluid damping (Damping through fluid and structure interactions)

4.2.1 MATERIAL (Internal) DAMPING

Material or internal damping of materials generally originates from the energy release associated with microstructure defects, such as grain boundaries and impurities; thermo elastic properties and effects can be utilised by local temperature gradients resulting from the non-uniform stresses, as in vibrating beams; eddy current effects in ferromagnetic materials; displacement motion in metals; and chain motion in polymers. Several simulations have been employed to represent energy release can be utilised by internal damping. This variety of models is primarily a result of the vast range of engineering materials; no single model can satisfactorily represent the internal damping characteristics of all materials.

4.2.1.1 ENERGY BALANCE APPROACH [50]

The loss factor η is commonly utilised to characterize energy release, due to inelastic behaviour, in a material subjected to cyclic loading. Assuming linear damping behaviour, η is defined by Vantomme [50] as;

$$\eta = \frac{1}{2\_\Pi} \frac{\Delta w}{w}$$

where Δw is the amount of energy released during the loading cycle and W is the strain energy stored during the cycle.

Now considering η1, η2 and η12 :
η1 – normal loading in fibre direction of UD lamina (longitudinal loss factor)
η2 – normal loading perpendicular to fibres (transverse loss factor)
η12—in plane shear loading (shear loss factor)

Two phase model

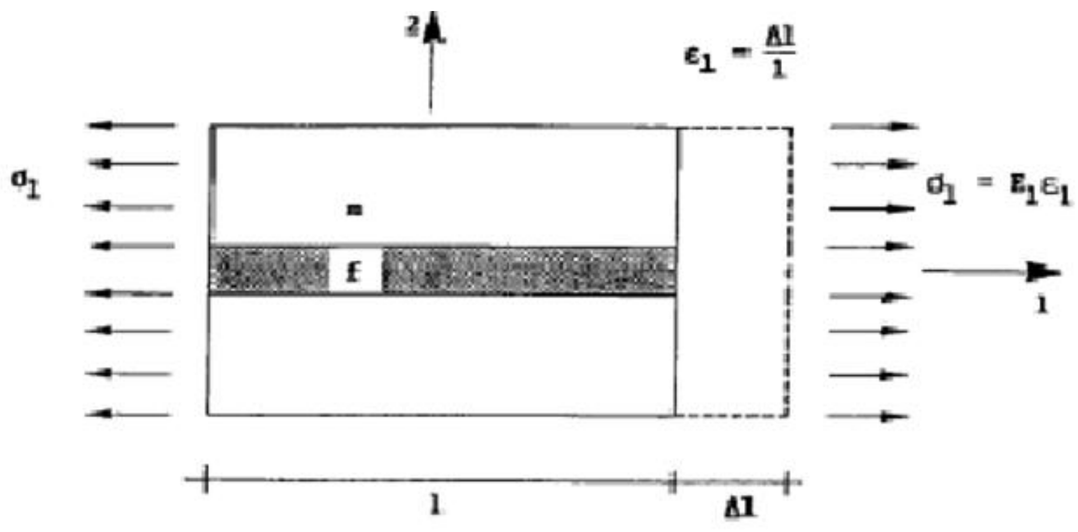


Fig.4.2 RVE loaded in 1-direction, Voigt model: matrix (m) and fibres (f) are connected in parallel

The longitudinal loss factor (η1) can be calculated by the following method: (loading in direction 1) the total energy released comprises the sum of that lost in the fibres and matrix. These amounts are proportional to the fraction of elastic strain energy stored in the fibres and matrix respectively;

i.e.,

$$\eta_1 = \eta_{fE}\frac{W_f}{W} + \eta_{mE}\frac{W_m}{W} \qquad \text{..........(1)}$$

ηfE and ηmE are the loss factors for fibres and matrix, associated with σ – ε tensile loading.

Using the expressions for the strain energy,

$$W_f = 1/2 \int_{V_f} \sigma_{1f}\epsilon_{1f}\,dV$$
$$W_m = 1/2 \int_{V_m} \sigma_{1m}\epsilon_{1m}\,dV \qquad \text{.......................(2)}$$

with

$$\sigma_{1f} = E_f\epsilon_{1f} = E_f\epsilon_1$$
$$\sigma_{1m} = E_m\epsilon_{1m} = E_m\epsilon_1 \qquad \text{.....................(3)}$$

gives:

$$W_f = 1/2\ E_f\epsilon_1^2\,V_f$$
$$W_m = 1/2\ E_m\epsilon_1^2\,V_m \qquad \text{...........................(4)}$$

Introduction of (4) into (2), with W = Wf + Wm, gives:

$$\boxed{\eta_1 = \frac{\eta_{fE}E_fV_f + \eta_{mE}E_mV_m}{E_fV_f + E_mV_m}} \qquad \text{.............(5)}$$

Transverse loss factor (η2) is calculated: (loading in direction 2)
As before, η2 is expressed as

$$\eta_2 = \eta_{fE}\frac{W_f}{W} + \eta_{mE}\frac{W_m}{W} \qquad \text{...................(6)}$$

The strain energy contributions are derived in an analogous manner as for η1, but now with the assumption that the same transverse stress σ2 is applied to both the fibres and the matrix. This development leads to

$$\boxed{\eta_2 = \frac{\eta_{fE} E_m V_f + \eta_{mE} E_f V_m}{E_m V_f + E_f V_m}} \qquad \text{............(7)}$$

Shear loss factor (η12) is calculated: (loading in shear direction)

$$\eta_{12} = \eta_{fG} \frac{W_f}{W} + \eta_{mG} \frac{W_m}{W} \qquad \text{..........................(8)}$$

where ηfG and ηmG are the loss factors for fibres and matrix associated with shear loading. The strain energy fractions are worked out in the same way as for η2, as it is assumed that the shear stresses on the fibres and matrix are the same. This leads to:

$$\boxed{\eta_{12} = \frac{\eta_{fG} G_m V_f + \eta_{mG} G_f V_m}{G_m V_f + G_f V_m}} \qquad \text{.................(9)}$$

Equation (9) indicates that damping for a UD lamina, for shear loading, is again matrix-dominated, because the stiffness Gf is usually much larger than Gm. The similarity of equations (9) and (7), combined with the fact that ηmE = ηmG, leads to the conclusion that η2and η12 should be very similar.

4.2.2 FLUID DAMPING

When a material is immersed in a fluid and there is relative motion between the fluid and the material, as a result the latter is subjected to a drag force. This force causes an energy release that is known as fluid damping.

The damping phenomenon can be applied to the machine tool systems in two ways:

1. Passive damping
2. Active damping

Passive damping refers to energy release within the structure by add on damping devices such as isolator, by structural joints and supports, or by structural member's internal damping.

Active damping refers to energy release from the system by external means, such as controlled actuator.

4.2.3 STRUCTURAL DAMPING

Rubbing friction or contact among different elements in a mechanical system causes structural damping [49]. Since the release of energy depends on the particular characteristics of the mechanical system, it is very difficult to define a model that represents perfectly structural damping. Coulomb-friction model is an imperative utilised to describe energy released due to friction. Regarding structural damping (energy released by contact or impacts at joints), energy release is determined by means of the coefficient of restitution of the two components that are in contact. Assuming an ideal Coulomb friction, the damping force at a join can be expressed through the following expression:

$$f = c.\text{sgn}\ (\dot{q})$$

where:

f = damping force, $\dot{q}$= relative displacement at the joint, c= friction parameter and the signum function is defined by:

$$\text{sgn}\ (x) = 1 \text{ for } x \geq 0$$
$$\text{sgn}\ (x) = -1 \text{ for } x < 0$$

4.3 DAMPING MECHANISMS IN COMPOSITE MATERIALS

Damping mechanisms in composite materials differ entirely from those in conventional metals and alloys [23]. The different sources of energy release in fibre-reinforced composites are:

a) Viscoelastic nature of matrix and/or fibre materials

(b) Damping due to interphase

(c) Damping due to damage which is of two types:

  (i) Frictional damping due to slip in the unbound regions between fibre and matrix.

  (ii) Damping due to energy release in the areas of matrix cracks and broken fibres etc.

(d) Damping in Viscoelastic materials.

(e) Damping in Thermo elastic materials.

CHAPTER 5
VISCOELASTIC DAMPING

5.1 FACTORS AFFECTING VISCO ELASTIC DAMPING:

Important viscoelastic behaviours that affect in damping are:

- Creep under constant stress
- Relaxation under constant strain
- Hysteresis loop due to cyclical stress
- Strain rate dependency on strain rate curve

These behaviours are discussed in the later sections of the chapter. This paper describes the damping behavior of carbon epoxy composite with riveted joint. The rivets utilised are mode of structural steel.

5.2 MATERIALS UTILISED

The composite material utilised in the analysis Carbon Fibre Composite Materials, Fibre / Epoxy resin (120°C Cure).

MECHANICAL PROPERTIES:

- Fibres @ 0° (UD), 0/90° (fabric) to loading axis, Dry, Room Temperature, V_f = 60% (UD), 50% (fabric)
- Epoxy resin and Standard CF Fabric

	Symbol	Units	Std CF Fabric
Young's Modulus 0°	E1	GPa	70
Young's Modulus 90°	E2	GPa	70
In-plane Shear Modulus	G12	GPa	5
Major Poisson's Ratio	v12		0.10
Ult. Tensile Strength 0°	Xt	MPa	600
Ult. Comp. Strength 0°	Xc	MPa	570
Ult. Tensile Strength 90°	Yt	MPa	600
Ult. Comp. Strength 90°	Yc	MPa	570
Ult. In-plane Shear Stren.	S	MPa	90
Ult. Tensile Strain 0°	ext	%	0.85
Ult. Comp. Strain 0°	exc	%	0.80
Ult. Tensile Strain 90°	eyt	%	0.85
Ult. Comp. Strain 90°	eyc	%	0.80
Ult. In-plane shear strain	es	%	1.80
Thermal Exp. Co-ef. 0°	Alpha1	Strain/K	2.10
Thermal Exp. Co-ef. 90°	Alpha2	Strain/K	2.10
Moisture Exp. Co-ef 0°	Beta1	Strain/K	0.03
Moisture Exp. Co-ef 90°	Beta2	Strain/K	0.03
Density		g/cc	1.60

Table 1: Properties of CARBON-FIBRE EPOXY Composite

CHAPTER 6

MATHEMATICAL MODELLING

6.1 STRUCTURAL DAMPING FACTOR (γ):

The viscous damping coefficient *c*, hysteretic damping coefficient *h* and the damping ratio ζ are considered to be the 3 important factors in damping of structures. But, there is another very vital factor, structural damping factor γ, to describe the property of the damping material.

The forced motion equation of a single spring mass system with a hysteretic damper is

$$m\ddot{x} + c_{eq}\dot{x} + kx = f(t)$$

For a harmonic problem, it becomes

$$-\omega^2 mx + k\left(1 - i2\frac{\omega}{\omega_n}\zeta_{eq}\right)x = f(t) \qquad \text{where } \zeta_{eq} = \frac{c_{eq}}{c_c} = \frac{h}{2m\omega_n\omega}.$$

For the modal damping, $\omega = \omega_n$, therefore, we have

$$m\ddot{x} + k(1 - i\gamma)x = f(t)$$

where, $\gamma = 2\zeta_{eq} = h/k$ is known as the structural damping factor or modal damping ratio.

For the viscous damping, similarly, the viscous damping factor is γ=2ζ.

6.2 COMPLEX STIFFNESS

The damping of the whole structure can be influenced by the polymer material due to its material stiffness as well as by its damping. These 2 properties are conveniently quantified by the complex Young's modulus or the complex shear modulus and E_η are usually assumed to be equal for a given material.

When the material is subjected to cyclic stress and strain with amplitude σ_0 and ε_0, the maximum energy stored and released per cycle in a unit volume are as

$$\text{Maximum energy stored per cycle} = E\varepsilon_0^2/2$$

$$\text{Energy dissipated per cycle} = \pi E\eta\,\varepsilon_0^2$$

A physical description of the loss factor can be found as follows. The energy released per cycle for a structural damped system is,

$$\Delta W = \pi hX^2 = \pi\eta kX^2 = 2\pi\eta\times\frac{1}{2}kX^2 = 2\pi\eta U_m$$

Where, U_m is the maximum strain energy stored. Therefore, we have energy strain maximum cycle per released energy

$$\eta = \frac{1}{2\pi}\frac{\Delta W}{U_m} = \frac{1}{2\pi}\frac{\text{energy disspated per cycle}}{\text{maximum strain energy}}$$

From the equation, it is found that the loss factor is a way to compare the damping of one material to another. It is a ratio of the amount of energy released by the system at a certain frequency to the amount of the energy that remains in this system at the same frequency. The more damping a material has, the higher the loss factor will be. The method of representing the structural damping should only be utilised for frequency domain analysis (modal) where the excitation is harmonic.

CHAPTER 7

MODELING AND ANALYSIS OF THE COMPOSITE WITH RIVETS

7.1 MODELLING:

As discussed earlier, the geometry and the structure of the composite material play an effective role in the reduction in damping. In this paper, a model was prepared using CATIA V5R17.The model prepared was a standard case in which 2 composite laminates were joined using a riveted joint and was discussed thoroughly. An assembled view of this model is shown below.

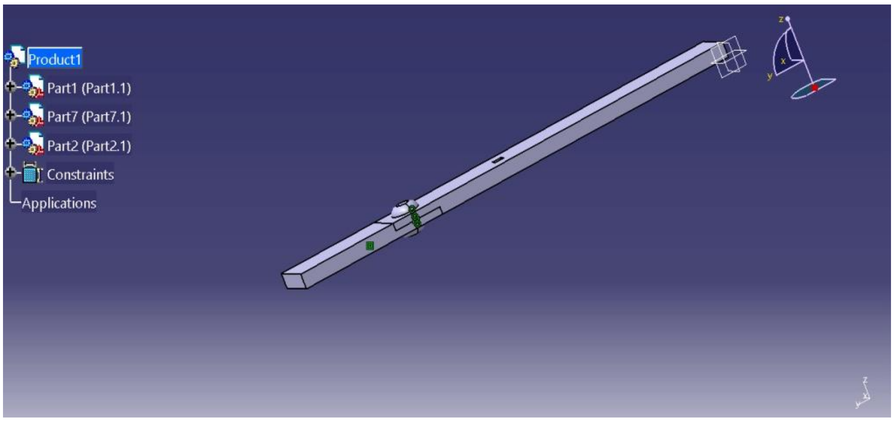


Figure 7.1: Model designed on CATIA V5R17

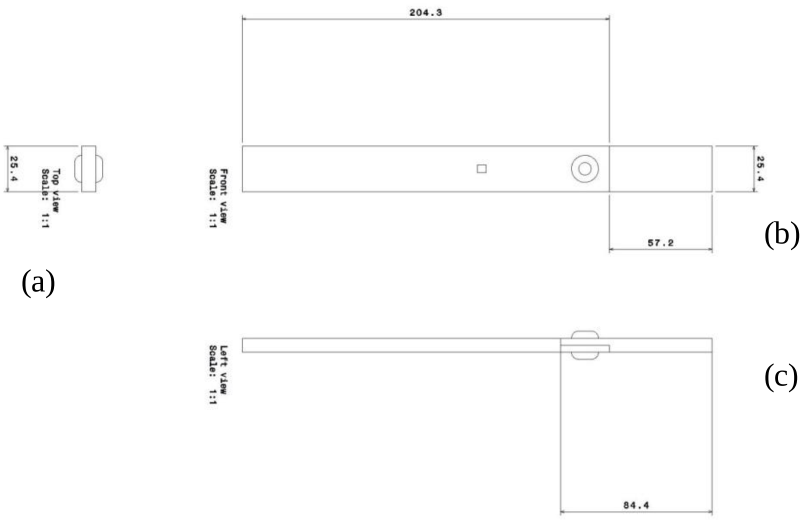


Figure 7.2: Various views of drafted models
(a) Front View, (b) Top View and (c) Side View of the drafted model.

7.2 ANALYSIS OF THE MODEL IN ANSYS

In this paper, using ANSYS software harmonic and modal analysis along with transient response and dynamic explicit modelling have been done for vibration damping. Several key points were deduced after the analysis of the model prepared.

7.2.1 GENERAL OVERVIEW OF DAMPING IN ANSYS

The damping matrix **C** in ANSYS may be utilised in harmonic, damped modal and transient analysis as well as substructure generation. In its most general form, it is:

$$[C] = \alpha[M] + \beta[K] + \sum_{j=1}^{N_{mat}} \beta_j [K_j] + \beta_c [K] + [C_\zeta] + \sum_{k=1}^{N_{ele}} [C_k]$$

Where,

α constant mass matrix multiplier

β constant stiffness matrix multiplier

βj constant stiffness matrix multiplier

βc variable stiffness matrix multiplier

$$\beta_c = \frac{\zeta}{\pi f} = \frac{2\zeta}{\omega} = \frac{\eta}{\omega}$$

ζ constant damping ratio, the damping ratio ζ should be 2η where η is the loss factor.

f frequency in the range between fb (beginning frequency) and fe (end frequency);

[Cζ] frequency-dependent damping matrix

[Cζ] may be calculated from the stated ζ_r (damping ratio for mode shape r) and is never clearly computed.

$$\{u_r\}^T [C_\zeta] \{u_r\} = 4\pi f_r \zeta_r$$

$\{u_r\}$ is the r^{th} mode shape

f_r frequency associated with mode shape r

$$\zeta_r = \zeta + \zeta_{mr}$$

ζ constant damping ratio

ζ_{mr} modal damping ratio for mode shape r

$[C_k]$ element damping matrix

7.2.2 RAYLEIGH DAMPING (α AND β):

The most common form of damping is the Rayleigh type damping.

$[C] = \alpha[M] + \beta[K]$.

In this representation, the matrix becomes the modal coordinates which is the major advantage of using this model.

$$\overline{\mathbf{C}} = \alpha\mathbf{I} + \beta\mathbf{\Lambda}$$

C' is the diagonal, so for the r^{th} mode, the equation of motion can be uncoupled. Each one is of the form

$$\ddot{q}_r + \left(\alpha + \beta\omega_r^2\right)\dot{q}_r + \omega_r^2 q_r = Q_r$$

Let $$2\zeta_{mr}\omega_r = \left(\alpha + \beta\omega_r^2\right)$$

The equation reduces to

$$\ddot{q}_r + 2\zeta_{mr}\omega_r\dot{q}_r + \omega_r^2 q_r = Q_r$$

Where, ζ_{mr} is the r^{th} modal damping ratio.

The values of α and β are not known directly, but are calculated from modal damping ratios, ζ_{mr}. It is the ratio of actual damping to critical damping for a particular mode of vibration, r. From the above equation, we have

$$\zeta_{mr} = \frac{\alpha}{2\omega_r} + \frac{\beta}{2}\omega_r$$

In many practical structural problems, the mass proportional damping α, represents frictional damping and may be ignored when (α = 0). In such case, the β damping can be estimated from known values of ζ_{mr} and ω_r which represents material structural damping. It is noted that only one value of β can be input in a load step, so we should select the most dominant frequency active in that load step to compute β.

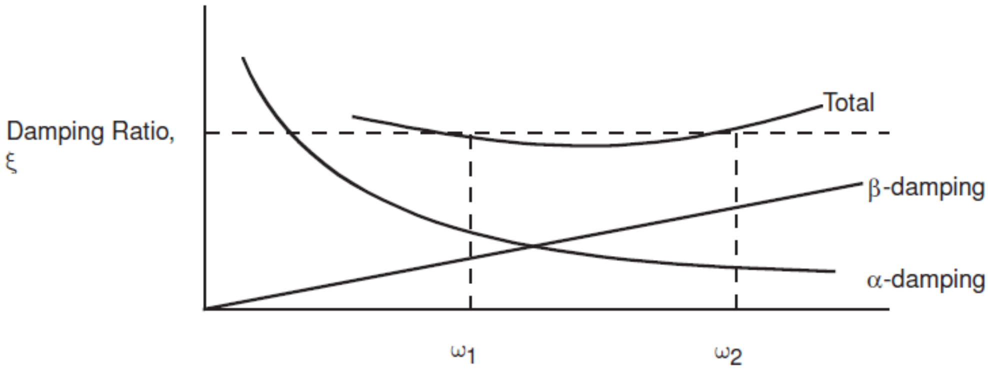


Figure 7.3: RAYLEIGH α and β DAMPING

CHAPTER 8

FINITE ELEMENTAL ANALYSIS OF THE MODEL

In this paper, various structural analyses have been done for the previously prepared model which was prepared in CATIA and then imported to ANSYS.

8.1 MODAL ANALYSIS

Modal analysis determines the natural frequency and mode shape of a structure. The natural frequency and mode shape are important parameters in the design of a structure for dynamic loading conditions and can be utilised in spectrum analysis or a mode superposition harmonic or transient analysis.

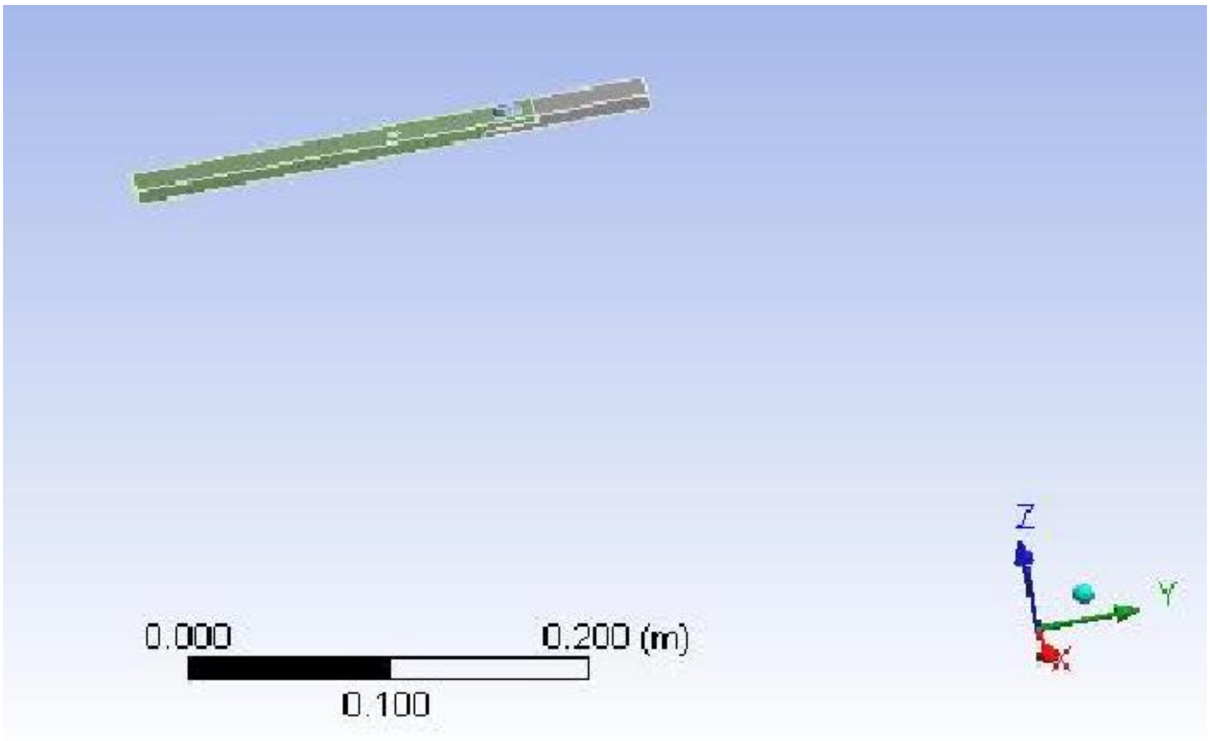


Figure 8.1: ANSYS Modal

Mode	Frequency [Hz]
1.	591.87
2.	1640.5
3.	1820.2
4.	3214.4
5.	3477.5
6.	4647.

Table 2: Frequency Output of modal analysis

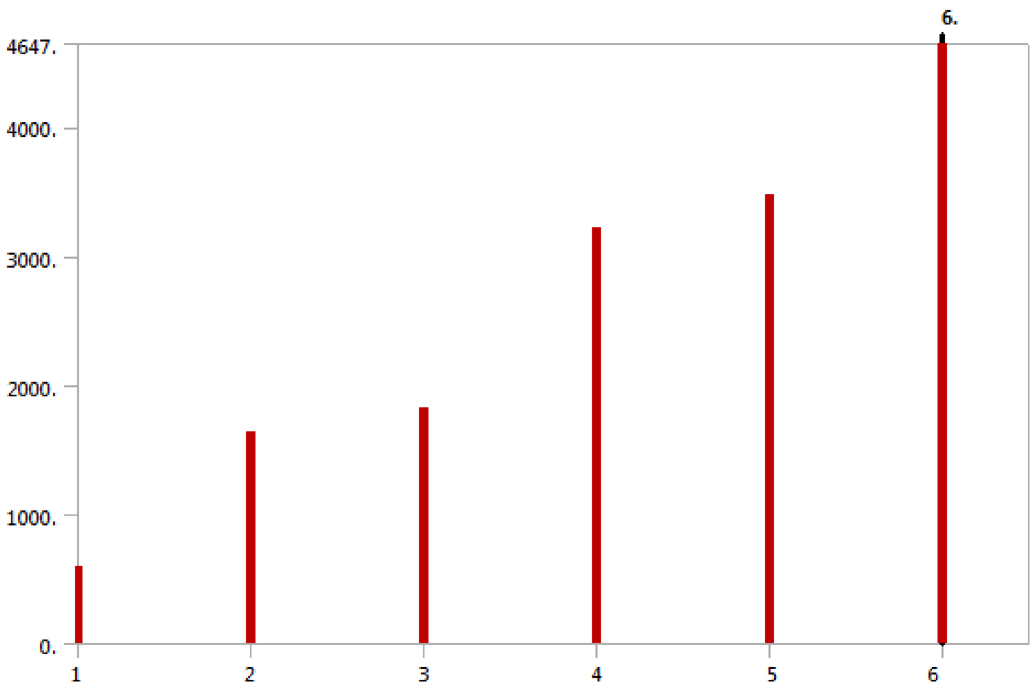


Figure 8.2: Mode Vs. Frequency graph

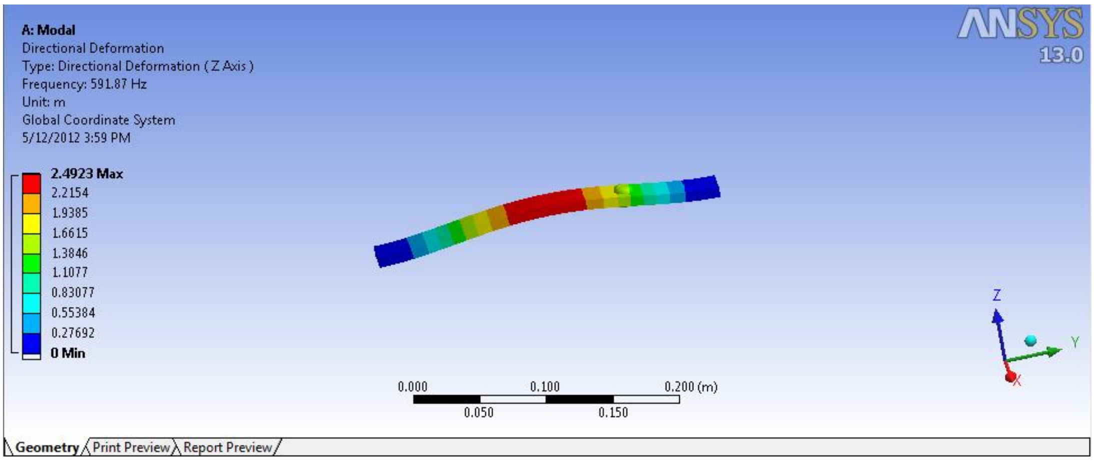


Figure 8.3: Modal analysis

8.2 HARMONIC RESPONSE ANALYSIS

It is a technique utilised to determine the steady state response of a linear structure to loads that varies sinusoidal with time. The mode superposition method calculations factored mode shapes (eigenvectors) from modal analysis to calculate the structures response. Hence it is known as harmonic response analysis.

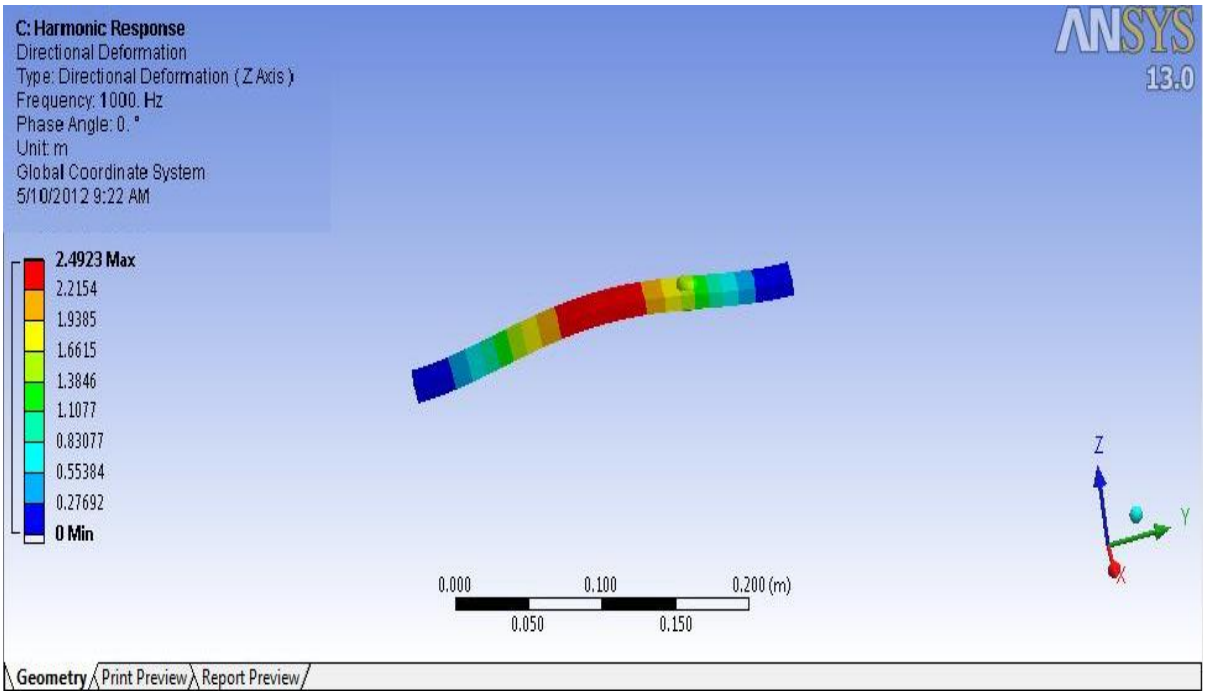


Figure 8.4: Harmonic response analysis of ANSYS model

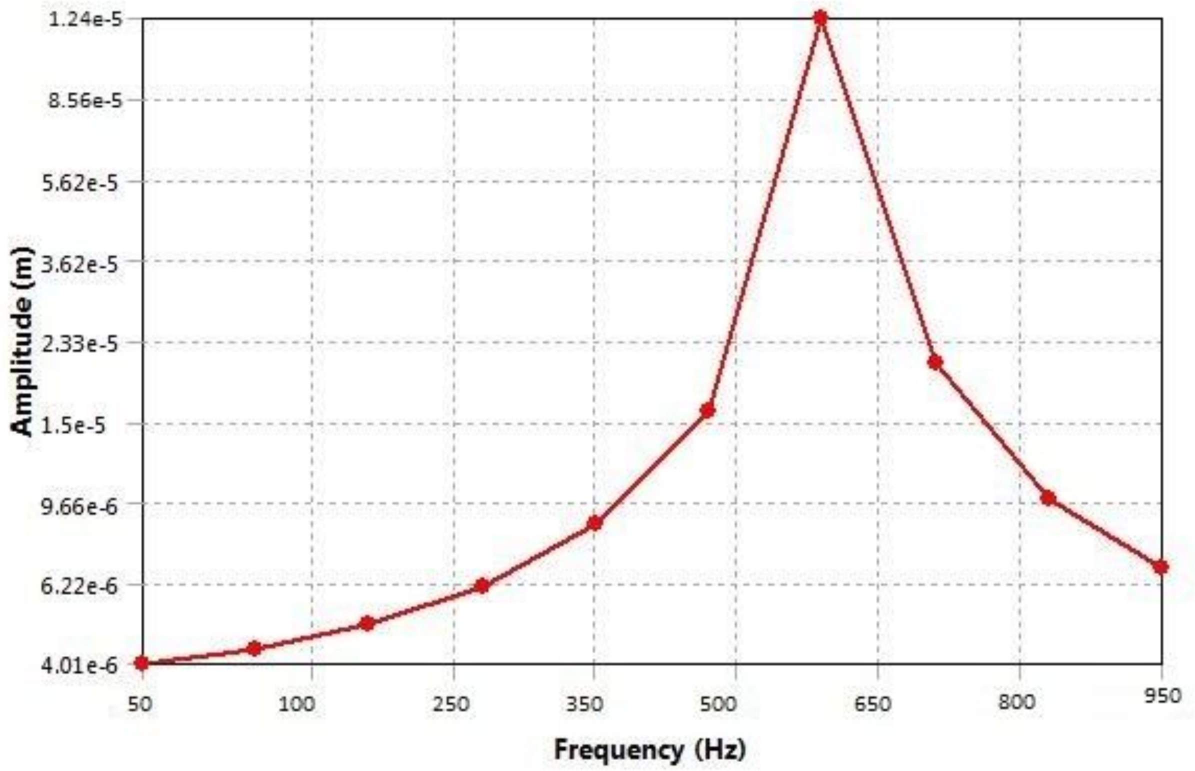


Figure 8.5: Amplitude vs. Frequency graph

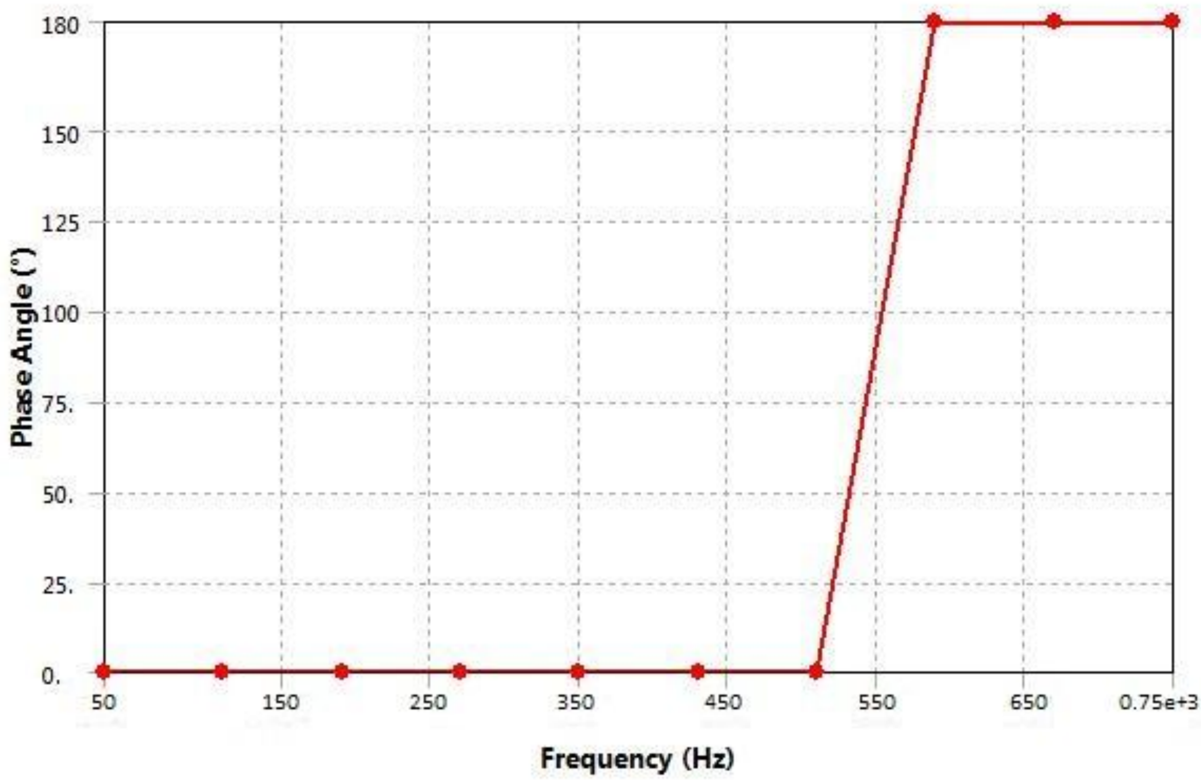


Figure 8.6: Phase Angle vs. Frequency graph

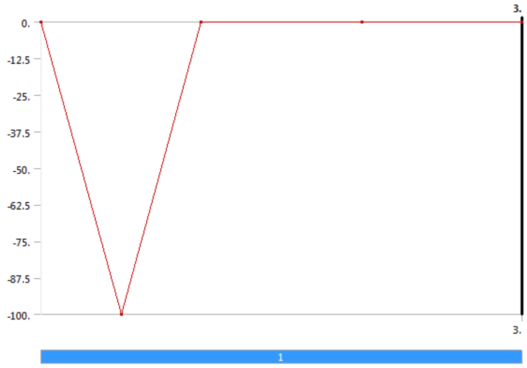
8.3 TRANSIENT DYNAMIC ANALYSIS

It is also called time history analysis. It is the technique utilised to determine the dynamic response of a system under the action of any time dependent load.

The basic equation of motion solved by a transient dynamic analysis is

$$(M)\{u''\}+(C)\{u'\}+(K)\{u\}=\{f(t)\}$$

Impulsive Load Input



Steps	Time [s]	Force [N]
1	0.	0.
	0.5	-100.
	1.	0.
	2.	
	3.	

The above graph shows how the impulse load is given to the structure. At time t=0.5 sec, an impulsive load of 100N is given to the structure. Analysis has been done to study the deformation of the structure and a graph is plotted between the min. deformation vs. time and maximum Deformation vs. Time. The tabular form of the table is given below:

Directional deformation along z axis in tabular form

2.4	Time [s]	Minimum [m]	Maximum [m]
2.5	0.1	-2.7126e-005	0.
2.6	0.2	-5.1285e-005	
2.7	0.3	-7.7733e-005	
2.8	0.4	-1.0366e-004	
2.9	0.5	-1.2941e-004	
3.	0.6	-1.0887e-004	
	0.7	-8.3124e-005	
	0.8	-5.6634e-005	
	0.9	-3.0069e-005	
	1.	-9.3384e-006	
	1.1	-7.9197e-006	9.7737e-010

	1.2	-7.9214e-006	9.0392e-010
	1.3	-7.9291e-006	9.9199e-010
	1.4	-7.9263e-006	9.0744e-010
	1.5	-7.9233e-006	9.8623e-010
	1.6	-7.9261e-006	9.132e-010
	1.7	-7.9235e-006	9.808e-010
	1.8	-7.9258e-006	9.2032e-010
	1.9	-7.9238e-006	9.7423e-010
	2.	-7.9256e-006	9.2631e-010
	2.1	-7.924e-006	9.6904e-010
	2.2	-7.9254e-006	9.3162e-010

Table 3: Maximum and Minimum deformation with time

2.3	-7.9242e-006	9.6432e-010

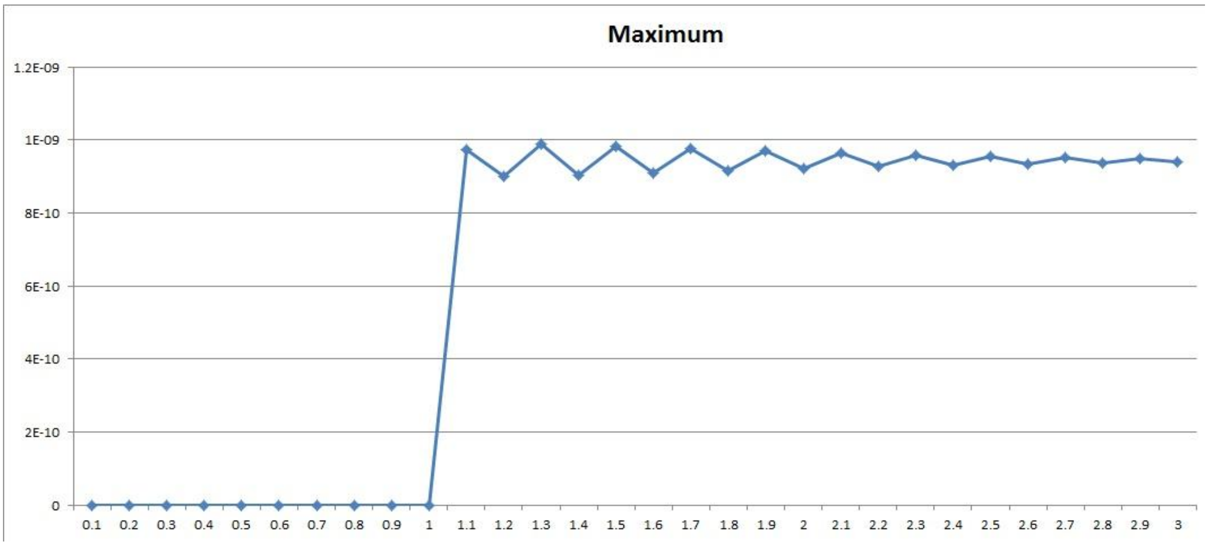


Figure 8.7: Maximum deformation vs. Time graph

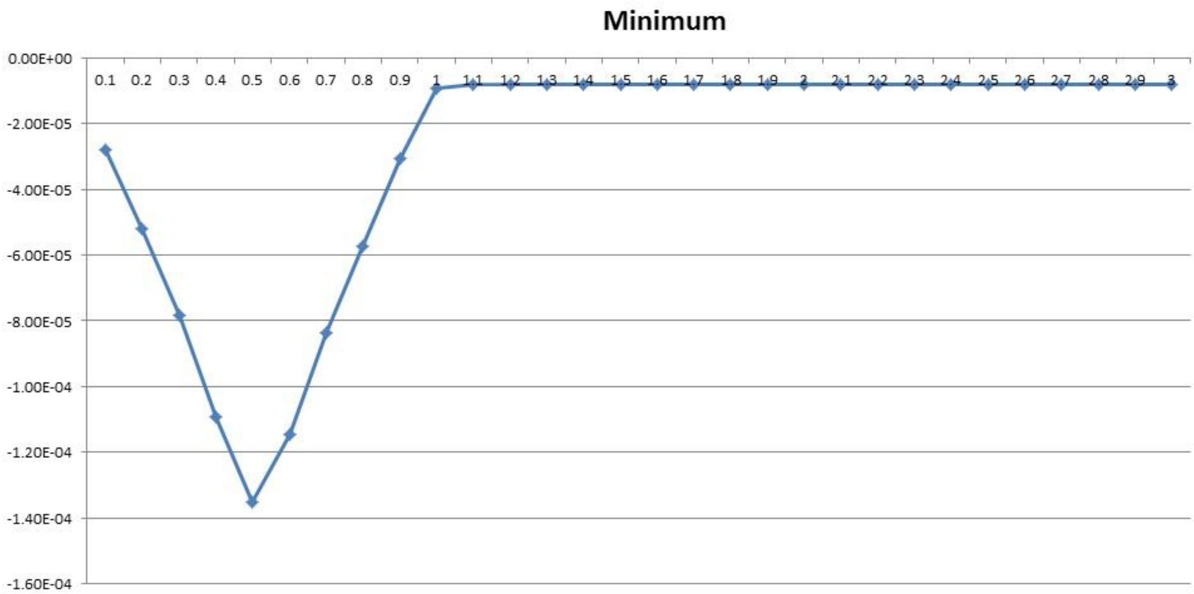
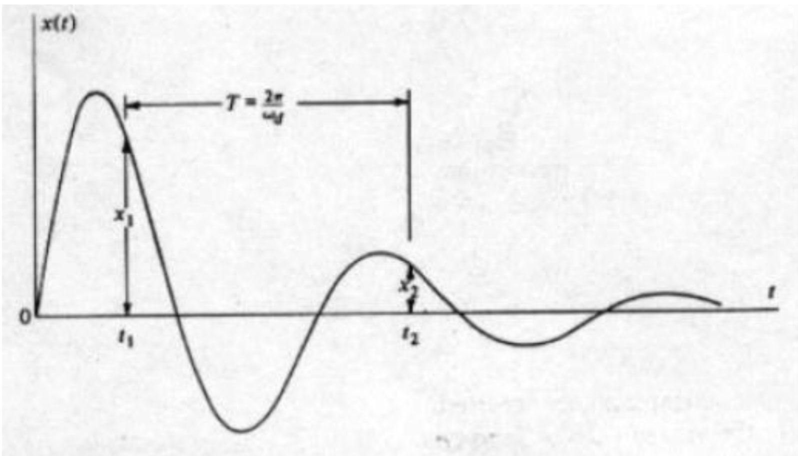


Figure 8.8: Minimum deformation vs. Time graph

CHAPTER 9

RESULTS AND CONCLUSIONS

- From modal analysis reported modal frequency = **591.87 Hz**
- From harmonic response model, Maximum strain energy = **8.68 X 10^{-5} J.**
- In transient analysis, the directional deformation along z axis with an impulsive force of **100 N** applied, the values of maximum deformation fluctuate and tend to converges to **9.42X 10^{-10}.**
- $\omega = 2\pi f$ = **3718.82 rad/sec.**
- logarithmic decrement, δ, as follows:



X1 and X2 are two consecutive displacements one cycle apart

δ **=** ln(x1/x2) = **6.3 X 10^{-3}**, X1 and X2 are taken from the values of the table

- $$\xi = \frac{\delta}{\sqrt{(2\pi)^2 + \delta^2}}$$

  ζ = 1.04X 10^{-3}.
- $\alpha = 2\zeta\omega$ = **7.471 s^{-1}** and $\beta = 2\zeta/\omega$ = **5.59 x 10^{-7}** s
- Energy released = $\pi c \omega x^2$ =**1.95 x 10^{-5}** J.
- Loss factor (η) =1/2π (energy released per cycle / maximum strain energy) = **0.0358**

REFERENCES

1. Cochardt, A.W., 1954, A method for determining the internal damping of machine members, ASME, Journal of Applied Mechanics, Vol. 76, No. 9, pp. 257-262.

2. Goodman, L.E. and Klumpp, J.H., 1956, Analysis of slip damping with reference to turbine-blade vibration, ASME, Journal of Applied Mechanics, Vol. 23, pp. 421–429.

3. Beards, C.F., 1992, Damping in structural joints, The Shock and Vibration Digest, Vol. 24, No. 7, pp. 3-7.

6. Ruzicka, J. E., “Structural Damping”. ASME, New Jersey (1959).

7. Lazan, B. J., “Damping of Materials and Members in Structural Mechanics”. Pergamon Press, London (1968)

8. Nashif, A. D., Jones, D. I. G. and Henderson, J. P., “Vibration Damping”. John Wiley & Sons, Inc., New York (1985).

9. Jones, D. I. G., “Handbook of Viscoelastic Vibration Damping”. John Wiley & Sons, Inc., New York (2001).

10. Sun, C. T., Lu, Y. P., “Vibration Damping of Structural Elements”. Prentice Hall PTR, Englewood Cliffs, NJ , (1995).

11. Beards, C. F., “Structural Vibration Analysis: Modeling, Analysis, and Damping of Vibrating Structures”. Halsted Press, Chichester (1983).

12. Ungar, E. E., "Structural Damping", Chapter 12 in Noise and Vibration Control Engineering: Principles and Applications, Leo. L. Beranek (Editor). John Wiley & Sons, Inc.,(1992).

13. Mead, D. J., “Passive Vibration Control”. John Wiley & Sons, New York (1999).

14. Bert, C.W., “Composite materials: A survey or the damping capacity of fibre reinforced composites in damping applications for vibration control”, AMD-Vol. 38. (1980).

15. Nashif, A.D, Jones, D.I.G and Henderson, J.P. Vibration Damping, Wiley, New York, (1985).

16. Chandra, R., Singh S.P, and Gupta.K Damping studies in fibre-reinforced composites—a review. Composite Structs, 46(1), 41-51, (1999).

17. Gibson, R.F., Chaturvedi, S.K. and Sun, C.T., “Complex moduli of aligned discontinuous

fibre-reinforced polymer composites", J. Mater Sci., Vol. 17. pp. 3499-3509 (1982).

18. Gibson, R.F., Chaturvedi, S.K. and Sun, C.T., "Internal damping of short-fibre reinforced polymer matrix composites", Comput. Struct 20, pp. 391—401, (1985).

19. Sun, C.T , Wu, J.K. and Gibson, R.F., "Prediction of material damping of laminated polymer matrix composits''. J. Mater. Sci. 22, pp. 1006-1012, (1987).

20. Adams, R.D. and Bacon, D.G.C, "Effect of fibre orientation and laminate geometry on the dynamic properties of CFRP , J Compos.Mater. 7, pp. 402-428, (1973).
21. Lin D.X, Ni R.G and Adams R.D, "Prediction and measurement of the vibrational damping parameters of carbon and glass fibre-reinforced plastics plates", J. Composite. Materials. 18, pp. 132 -152 (1984).

22. Ni R.G and Adams R.D., "A rational method for obtaining the dynamic mechanical properties of lamina for predicting the stiffness and damping of laminated plates and beams", Composites 15, pp. 193—199, (1984).

23. Morison W.D, "The prediction of material damping of laminated composites", Can. Aeronaut. Space J. 28, pp. 372-382, (1982).

24. Kinra, V.K., Wren G.G., Rawal S.P. and Misra. M.S. "On the influence of ply-angle on damping and modulus of elasticity of a metal-matrix composite", Metall. Trans, 22A. pp. 641- 651. (1991).

25. Ungar, E.E. and Kerwin, J.R., "Loss factors or viscoelastic systems in terms of energy concepts," J Acoust. Soc. Am., 34, No. 7, 954.958, (1962).

27. Johnson C. D. and Kienholz D. A., "Finite element prediction of damping in structures with constrained Viscoelastic layers," AIAA J., 20, No. 9, 1284-1290,(1982).

28. Gibson, R. F. and Plunkett R. , "A forced-vibration technique for measurement of material damping. Experimental Mechanics", Volume 17, Number 8 / August, : 297-302, (1977).

29. Suarez, S. A., Gibson, R. F. and Deobold, L. R., "Random and Impulse Techniques for Measurements of Damping in Composite Materials," Experimental Techniques, Vol. 8, No. 10, pp. 19–24,(1984).

30. Gibson Ronald, F. "Modal vibration response measurements for characterization of composite materials and structures". Composites science and technology, American Society of Mechanical Engineers, Aerospace Division's Symposium (1999) 2000, vol. 60,pp. 2757-2802 (42 ref.), pp. 2769-2780.

31. Lin D.X, Ni R.G, Adams R.D. "Prediction and measurement of the vibrational damping

parameters of carbon-glass fibre-reinforced plastic Plates". J Comp Mater; 18: 132-52.(1984).

32. Koo K.N, Lee.I. "Vibration and damping analysis of composite laminates using deformable finite element". AIAA J; 31 (4): 728-35,(1993).

33. Singh S.P., Gupta K., "Damped free vibrations of layered composite cylindrical shells". J Sound Vib 172 (2) : 191-209 (1994).

34. Myklestad N. O., "Journal of Applied Mechanics. The concept of complex damping". 19, 284-286,(1952).

35. Corsarov R.D., Sperling L.H., "Sound and Vibration Damping with Polymers", ACS Symposium Series, Vol. 424, American Chemical Society, Washington, DC, (1990).

36. Madigosky, W.M., Lee G.F., "Improved resonance technique for materials characterization", Journal of the Acoustical Society of America, 73, 1374–1377, (1983).

37. Rahman, M., Mansur, A. and Karim, B. "Non-conventional material for machine tool structures"[J]. JISM International Journal, Series C, 44(1), 1–11, (2001).

38. Lee, D.G., Chang, S.H. and Kim, H.S. "Damping improvement of machine tool columns with polymer matrix fibre composite materials". Compos. Struct., 43,155–163, (1998).